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STRENGTH ANALYSIS OF PROPELLER SHAFTING ON ORCA CLASS FISHERIES INSPECTION BOAT USING FINITE ELEMENT METHOD

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ABSTRACT

Propeller shafting system is machinery elements which important for ship propulsion. The function of propeller shafting is to distribute the power from the main engine or main gearbox to shaft propeller. The rigid coupling system is a type of shafting system which is simpler and most used in ship machinery. The disadvantage of this coupling is not permitting any misaligned in the mids of coupling to shaft end and cannot reduce the transmission impact. The torsion that taken out by main engine will be causing failure or breaking the coupling, particularly on key and bolt. Damage in coupling is a major problem because the main engine cannot distribute the power to the propeller. Thus, the ship cannot be able to move. The goal of this investigation is known the proper material with considering the yield strength of materials. Finite Element Method is a numerical simulation which not needed a real test. Contact mechanics are considered for the bearing frictions which the shaft using the water-lubricated seal, which required a durable material against corrosion. According to the results, the maximum stress which is obtained located on keyway element when maximum continuous rating condition. This condition should be damaged the shaft and key Meanwhile the maximum stress which is obtained under 75% of engine rating condition was in safety limit. Thus, UNS 41600 is a superior choice for aluminum used parts.

Key words: Propeller Shafting, Rigid Coupling, Contact Mechanics, Water-Lubricated Bearings.

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1. INTRODUCTION

Indonesia is in 3rd world place of fisheries production. Due to Indonesia have large fisheries resources. This thing makes Indonesian waters are vulnerable to be illegal fishing target. Therefore, the fisheries inspection boats are necessary as law enforcement duties. The fisheries inspection boat must have high endurance, due to the high operational hour. That is an important thing to planned the material selection, particularly on propeller shafting.

Dymarski and Narewski have performed an analysis of ship shaft line coupling bolts failure. According of their analysis, cause of the failure is excessive wearing of bearings, accelerated stern bearing clearance growth and shaft vibrations. ^[1] Pantazopoulos and Papaefthymiou were performed a fracture analysis of austenitic stainless steel propeller shaft. They were using SEM (scanning electron microscope). The fracture of austenitic stainless steel which used for shaft is caused by misaligned installation, excessive clearance and vibration. ^[2] Jaiswal and Pasarkar have analyzed the failure of flange coupling for industrial machinery. The design criteria for flange coupling is consider by stiffness of material, RPM and environmental factors. ^[3] Jose Ananth and Hameed have performed an analysis by comparing the steel shaft propeller with the composite of E-Glass fiber reinforced plastic shaft. The usage of composite material can be reduce the weight of shaft. ^[4] Litwin was investigated an experiment of the properties of propeller shaft with water-lubricated bearing. Comparison between water-lubricated bearing with oil-lubricated that is the water-lubricated bearing is more environmental friendly as no risk of pollution. ^[5]

The failure of machinery elements on the propulsion system usually occurred, sometimes during ship cruise. Particularly the failure on propeller shafting system. Such as, broken bolts of coupling and the key. To prevent failure on propeller shafting, therefore a numerical simulation is necessary. The simulation also considering the contact mechanics of each component, for instance the friction of bearing. The goal of this research is to obtain the von mises stress, deflection value and safety factor using finite element method. Then, finding the appropriate materials for each component of propeller shafting.

1.1. Rigid Coupling

Rigid coupling used if the ends of the shaft are axially connected. The flange coupling fig 1 consists of the flange hub, which is made from cast steel or forged steel, and assembled at the end of shaft with key, then fastened with bolts. ^[6]

This coupling not allowed any misaligned on each end of the shaft and cannot reduce any vibration of transmission. At it installation, each end of the shaft must be attempted in a line before the bolts of flange have fastened.

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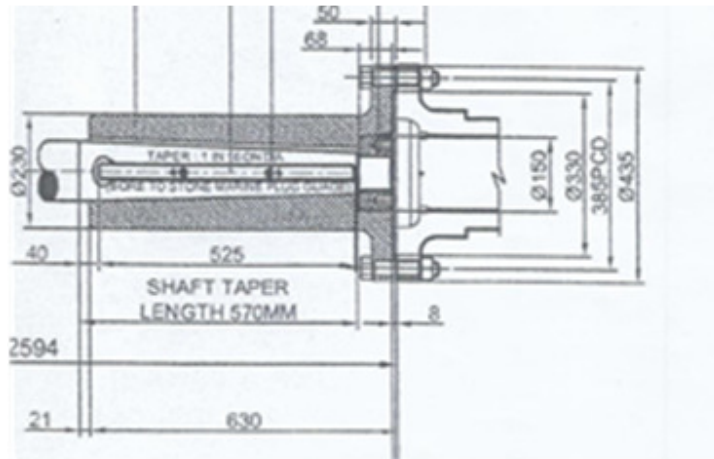


Figure 1 Rigid Flange Coupling

1.2. Torsional Shear Stress

Based on coupling mechanism which is a rotational motion, therefore to calculate the stress of coupling must be using the torsion theory. According Popov, E, the shear stress expressed as, [7]

$$\tau_{\max} = \frac{Tr}{J} \quad (1)$$

The calculation of moment of inertia is distinguished to be two type, that is the solid fig 2 and hollow cylinder fig 3 expressed as,

$$J = \int \rho^2 dA = \int_0^c 2\pi \rho^3 d\rho = 2\pi \left[\frac{\rho^4}{4} \right]_0^c = \frac{\pi c^4}{2} = \frac{\pi d^4}{32} \quad (2)$$

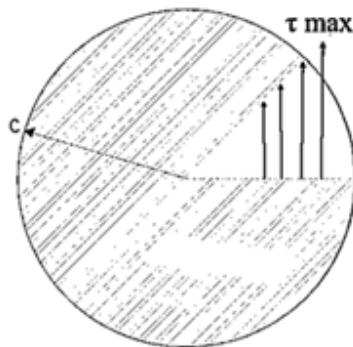


Figure 2 The Inertia Variation on Solid Cylinder

whereas for moment inertia on hollow cylinder is,

$$J = \int \rho^2 dA = \int_0^c 2\pi \rho^3 d\rho = \frac{\pi c^4}{2} - \frac{\pi b^4}{2} \quad (3)$$

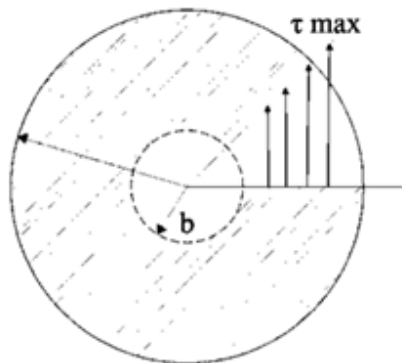


Figure 3 The Inertia Variation on Hollow Cylinder

1.3. Failure Criterion

There are the various stress if a structure was charged by load. It is depended by the vector of load and the behavior of material. For the instance, there are axial stress and shear stress if a structure charged by compressive load. Likewise the torsional load, however the axial stress was smaller. The impact of structure under load, it can be change the shape and volume of structure. There are some theory which is studied about maximum distortion energy as M. T Hueber and R. Von Mises.^[8] To calculate the stresses on the problem of three dimensional material. It more convenient if using equivalent stress (σ_e). Defined as the value of uniaxial tensile stress that would produce the same level of distortion energy as the actual stress involved. The equivalent stress equation expressed as,

$$\sigma_e = \frac{\sqrt{2}}{2} [(\sigma_2 - \sigma_1) + (\sigma_3 - \sigma_1) + (\sigma_3 - \sigma_2)^2]^{\frac{1}{2}} \quad (4)$$

where σ_n is principal stress by the cross section of axis. It can be derived depend on each cross section, the equivalent stress equation is,

$$(\sigma_x^2 + \sigma_y^2 - \sigma_x\sigma_y + 3\tau_{xy}^2)^{\frac{1}{2}} \quad (5)$$

As shown in fig 4, the failure of material is limiting by it yield strength. To knows the elastic limit of material, we have Von Mises circle and Tresca hexagon. Assumed that σ_1 is 1st principal stress and σ_2 is 2nd principal stress. Thus, those for each principal stresses cannot be exceeds the yield strength of material (σ_y).

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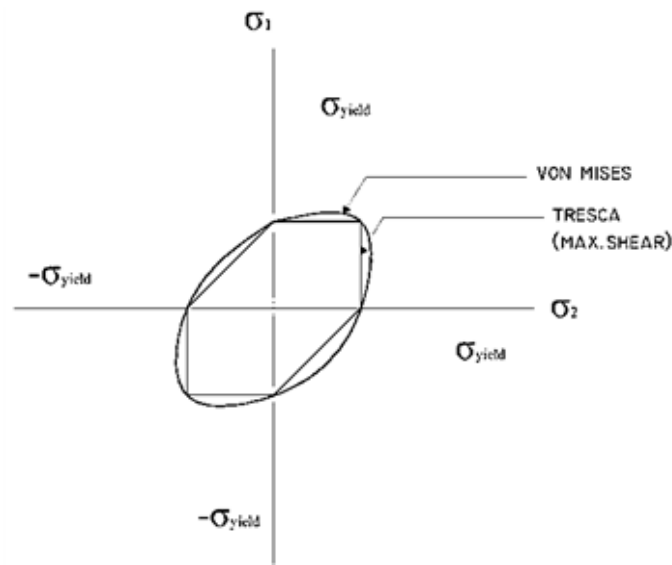


Figure 4 Von Mises Circle and Tresca Hexagon

1.4. Safety Factor

Safety factor is a ratio of structure capacity under given load. Safety factor is used to Calculate the strength on a structure design. To reach safety factor, the yield strength of material must be larger than working stress. According to Young, W, safety factor is a fractional number of the structure's highest capability which is necessary to withstand a load.^[9]

$$\text{Factor of safety} = \frac{\text{Yield Strength}}{\text{Working Stress}} \quad (6)$$

A structure with a specific safety factor will withstand only the load which is planned. Each material has different resistance, therefore the ratio of safety factor obtained by comparison between yield stress against working stress on the structure.

2. DATA USED AND MATERIAL SELECTION

The investigation of propeller shafting consisted of three important components. Thus, this section discusses the material properties, the torsional load, and the contact mechanics.

2.1. Material Properties

Material selection is an important part of planning the machinery components. The material has different properties and has a different function. In this case, Molybdenum steel (AISI 4340) is used for the flange. This steel contains about 15% to 30% of molybdenum (Mo). Containing 0.4% carbon and high strength martensitic steel which is desirable and is used very frequently to manufacture critical components in aerospace engineering and automotive transmission.^[10] Properties of AISI 4340 are given in table 1.

Stainless Steel 2205 Duplex is used for shaft and key. The duplex structure improves stress-corrosion crack resistance.^[11] Properties of Stainless Steel 2205 Duplex are given in table 2. For bolts and nuts are using AISI 316 Stainless Steel.

AISI 316 steel presents outstanding intergranular corrosion resistance, good for grain corrosion resistance to most chemicals, salts and acids. Also Mo content helps increase resistance to marine environments.^[12] Properties of AISI 316 are given in table 3. The advantages of stainless steel are have good resistance against corrosion and can be used for watery environment. Because, the shafting system using sea water lubricant.

Table 1 Material Properties of AISI 4340

AISI 4340		
Yield Strength	470	Mpa
Ultimate Tensile	745	Mpa
Density	7850	Kg/m ³
Elastic Modulus	210	Gpa

Table 2 Material Properties of 2205 Duplex Stainless Steel

2205 Duplex Stainless Steel		
Yield Strength	448	Mpa
Ultimate Tensile	621	Mpa
Density	7820	Kg/m ³
Elastic Modulus	190	Gpa

Table 3 Material Properties of AISI 316 Stainless Steel

AISI 316 Stainless Steel		
Yield Strength	205	Mpa
Ultimate Tensile	515	Mpa
Density	8000	Kg/m ³
Elastic Modulus	193	Gpa

2.2. Calculating the Torsional Load

To calculate the load, given the engine brake horse power, angular speed and the gearbox ratio as shown in table 4. This is the specification of main engine and main gearbox,

Table 4 Main engine and gearbox specification

Engine and Gearbox Specifications ^{[13],[14]}		
Power	2560	kW
Angular Speed	1970	RPM
Gearbox Ratio	4.522:1	

For the first step, finding the output of gearbox angular speed,

$$\begin{aligned}
 n_{\text{output}} &= n_{\text{input}} / \text{ratio} \\
 &= 1970 / 4.522 = 435.64 \text{ RPM}
 \end{aligned}$$

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Then calculating the torsional load, from equation below,^[15]

$$T = \frac{P \times 60 \times K}{2\pi n} = \frac{2560 \times 60 \times 1000}{2 \times 3.14 \times 435.64} = 56144.06 \text{ N.m} \quad (7)$$

where K is 1000 if using metric unit.

2.3. Contact Mechanics

Due to each components are independently assembled, contact definition is needed. In this case, each components must be use penalty contact. In the real operation, friction between component surfaces were occurred on propeller shafting system. Hence, the friction coefficient should be attached on contact definition. Each components on propeller shafting system have a different friction coefficient. For contact between steel-ThorPlas, according G.Ren and J. Feng the ThorPlas which contacting the shaft at wet condition is about 0.1 to 0.17.^[16] And the friction coefficient between steel-steel is 0.8 at dry condition for contact between bolt, nut, coupling and key.

3. RESULTS AND DISCUSSION

The result was performed by Finite Element software, in this case Solidworks Simulation was used. The Von Mises stress (σ_v) and resultant deformation (u_{total}) were calculated to found the behavior of those material, considered by friction coefficient with Thor-Plas bearing. The axial stresses assumed zero, due to the load is torsional. To know the material still in safety limit. The safety factor was calculated as a parameter of material selection.

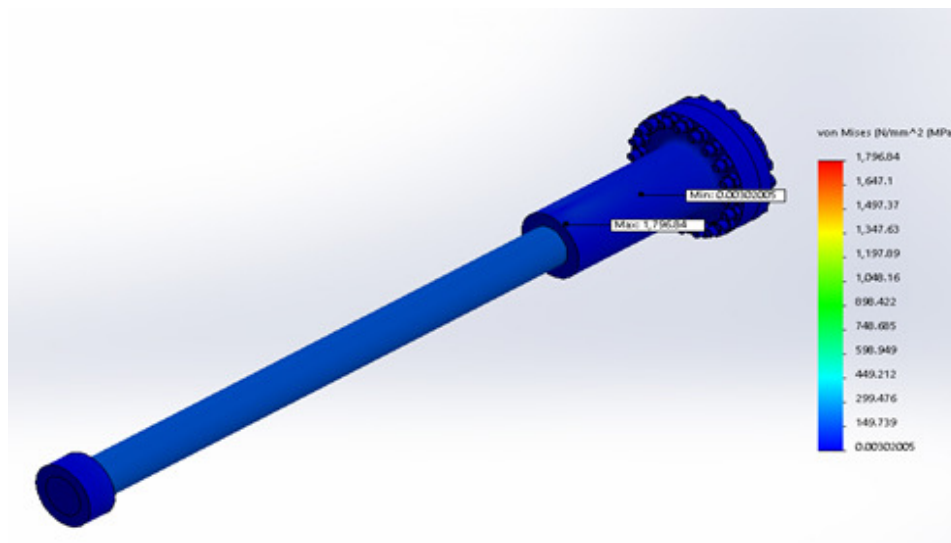


Figure 5 Maximum Von Mises Stress

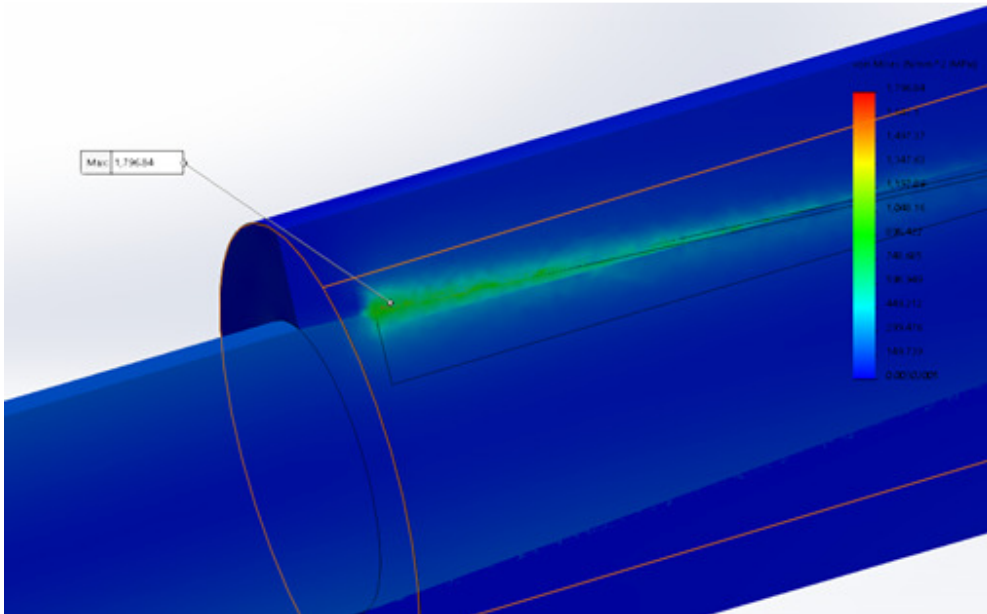


Figure 6 Detailed Maximum Von Mises Stress on cut-section of Key

As shown in fig. 5 and 6, the value of maximum Von Mises stress which located on key component, the value of stress is 1796.84 N/mm². While, minimum Von Mises stress which located on female flange which the value of stress is 3.02E-3 N/mm². Where, the axial stress can be assumed by zero due to the torsional load. Thus, for the stress we have torsional shear stress.

This stresses value are obtained at maximum continuous rating of main engine, which about 110% rating and maximum torque.

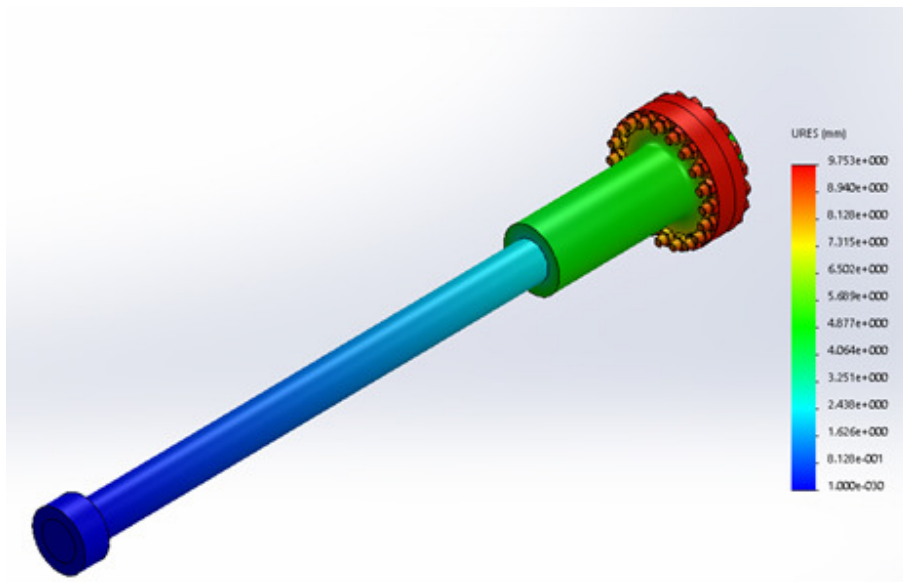


Figure 7 Maximum of Total Displacement

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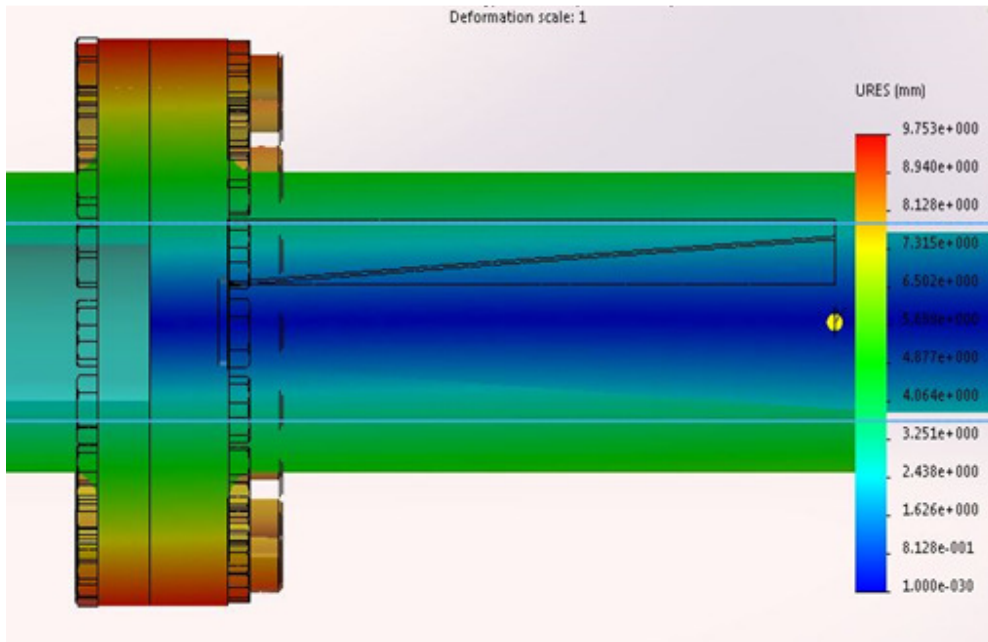


Figure 8 Maximum of Total Displacement on cut-section of Key

As shown in figure 7 and 8 the maximum displacement is located on the edge of flanges, which have 9.753mm of displacement value. The minimum value is located on the end of after shaft, which have 0.812mm. In this case, the displacement values are obtained from total displacement. Due to torsional load, suppose assumed that the axial displacement value is zero. Thus, the rotational displacement will be used. Although, the axial displacement also occurred, however the value is too small.

From the simulation above, obtained the working stress of each components. Thus, the yield strength of components can be compare to working stress which is occurred to obtained the value of safety factor. In this simulation we have two conditions of main engine rating. The calculation of safety factor can be shows in table 5 and 6,

Table 5 The Value Of Safety Factor at 110% Rating of Main Engine

Component	Max. Working Stress (N/mm²)	FoS
<i>Male Flange</i>	120.05	3.9
<i>Female Flange</i>	1380.3	0.3
Key	596.52	0.8
Shaft	1796.8	0.2
Bolts and Nuts	49.756	4.1

Table 6 Value of safety factor at 75% rating of main engine

Component	Max. Working Stress (N/mm²)	FoS
<i>Male Flange</i>	53.496	8.8
<i>Female Flange</i>	366.95	1.3
Key	285.48	1.6
Shaft	538.7	0.8
Bolts and Nuts	24.675	8.3

According the BKI: Part. I Vol.III “Rules for Machinery Instalation”.^[17] For the value of safety factor which is applied on propeller shafting system (not between main engine to gearbox), obtained the limit value 2.5.

4. CONCLUSION

Based on the field data. The engine load condition is divide by 110% and 75% of continous rating. It is considered because of the ship used the engine for cruise between 110% to 75% of engine rating. The conclusions can be mentioned as,

1. The maximum Von Mises stress which obtained is located on key and keyway (part of shaft). Which the stress value is 1796.84 N/mm² at 110% rating of main engine and 538.7 N/mm² at 75% rating of main engine. Thus, it can be expressed that the propeller shafting system still safe during the operational cruise if the rating of main engine is not exceed 75%, except at short operational for instance during sea trial.
2. The minimum safety factor of each components is located on the keyway (part of shaft), that is 0.8.
3. According the rules of BKI: Vol.III “Rules for Machinery Instalation”, the value of safety factor is not exceed 2.5. Therefore, the material selections which is adviced is has higher yield strength. For the shaft is adviced to using Stainless Steel UNS 41600, which is have 1050 n/mm² of yield strength.

Thus, from the result of finite element simulation, can be obtained the Von Mises stress and deflection. Then, defining a material which is compatible.

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